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GASES AS SECONDARY HEAT TRANSFER FLUIDS

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DEPARTMENT OF ENGINEERING RESEARCH
NORTH CAROLINA STATE COLLEGE
RALEIGH, NORTH CAROLINA

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Report prepared by: J. S. Doolittle
W. O. Doggett
C. F. Martin

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A. C. Menius, Jr.
Technical Director

J. S. Doolittle
Associate Technical
Director

ABSTRACT

Alkali liquid metals, because of their superior heat transfer characteristics, rate highly as secondary heat transfer fluids for systems consisting of a high temperature secondary heat exchanger, a transfer piping system, and one or more air radiators.

However, because alkali liquid metals are highly corrosive when even very small amounts of impurities are present, consideration ~~should be~~ is given to ~~the use of other fluids in the secondary loop~~. In this report the relative merits of gases. ~~are examined.~~ ~~Since hydrogen is an excellent heat transfer gas, it was used throughout this study.~~

The relative effect of the use of hydrogen rather than an alkali liquid metal is presented for the secondary heat exchanger, the transfer piping, and the air radiator.

In general, by a change in the configuration of the component parts, it is possible when using hydrogen to reduce weights to values comparable with those required for the liquid alkali metals. However, gas pressures must be in the order of several thousand pounds per square inch and pumping powers used which are several times those for the liquid alkali metals. Similarly, by using larger parts whose weights are several times those required for the alkali liquid metals, the pumping horsepowers may be reduced to values similar to those for the liquid alkali metals.

On the other hand, hydrogen at pressures at least 1,000 pounds per square inch is superior from the weight-pumping power requirements to the heavy liquid metals such as lead-bismuth. Furthermore, when weight is not a prime consideration, hydrogen shows considerable promise as a secondary heat transfer fluid.

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I. INTRODUCTION

As with previous reports on this project (1, 2, 3),[1] it is assumed that the system under consideration consists of a high temperature heat source, a secondary heat transfer loop, and one or more gas turbines located in close proximity. Heat is transferred in the heat source to a primary fluid which is conveyed to a secondary heat exchanger, located adjacent to the heat source. A secondary fluid, heated in the secondary heat exchanger, is transported to the air radiators where, in turn, it heats the air before entrance to the gas turbines.

In many respects, the alkali liquid metals are the best secondary heat transfer fluids. Their thermal film resistance is extremely low. Their density is lower than that of most other liquids. For these reasons, secondary systems using an alkali liquid metal are compact and have a low total weight. Pumping power requirements are of reasonable values. However, the alkali liquid metals are highly corrosive, particularly when any impurities, even in minute amounts, are present. Thus it is a real problem to find containing metals, especially at the elevated temperatures encountered in secondary heat transfer loops designed for maximum heat transfer.

Gases have certain advantages for this type of service. Most of them are far less corrosive and hence are easier to

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[1] See Bibliography for titles of these reports.

contain than the alkali liquid metals. Even under high pressures, the density of gases is only a fraction of that of liquids. (The fluid density is of prime importance when long transfer pipes must be used between the heat source and the air radiator.) On the negative side, gas films offer a high resistance to heat transfer, thus requiring larger heat exchangers than do the liquid metals. In addition, because of the low density of gases, they require large flow areas and pumping power requirements are large, as is the weight of the piping system.

It is the purpose of this report to examine, in a quantitative manner, the advantages and disadvantages of gases as secondary heat transfer fluids and to present information from which the relative merits of gases as secondary heat transfer fluids may be ascertained. The approach to be followed will be to consider separately the three parts of the secondary loop; namely, the secondary heat exchanger, the transfer piping, and the air radiator.

II. SECONDARY HEAT EXCHANGER

The same general assumptions that were made in previous reports (1, 2, 3), relative to the operating conditions of the secondary heat exchanger, will be made here. These are:

1. The heat transferred is equivalent to 100 megawatts (94,800 Btu/sec).

2. The temperature of the primary fluid decreases from 1550°F to 1050°F in the secondary heat exchanger.

3. The secondary fluid is heated from 1000°F to 1500°F.

4. The heat exchanger is single pass-counterflow, with the primary fluid in the shell side.

Because the primary object of this report is to establish whether or not gases have sufficient qualifications to justify a detailed study of their merits relative to other fluids, the following restrictions will be assumed:

1. The primary fluid is sodium.

2. The tubes are type 316 stainless steel.

3. The ratio of inside to outside diameter of the tubes is 0.84.

The three quantities of heat involved in the secondary heat exchanger are the heat given up by the primary fluid, the heat transferred through the tubes, and the heat received by the secondary fluid. Neglecting losses to the surroundings, these three quantities must be equal. Expressing these quantities in equation form,

$$Q = M_o C_o \Delta t_o \quad (1)$$

$$Q = U A_t \Delta t_m \quad (2)$$

$$Q = M_i C_i \Delta t_i \quad (3)$$

where Q = heat flow rate, Btu/hr

M = mass flow rate, lb/hr

C = constant pressure specific heat, Btu/lb-°F

U = over-all heat transfer coefficient, Btu/hr-ft^2-°F

Δt = temperature difference, °F

and subscripts i, o, and m signify inside (tube side), outside (shell side), and logarithmic mean,[2] respectively.

The heat transfer area, based on the shell side of the tubes, is

$$A_t = \pi D_o L N \quad (4)$$

where N = the number of tubes

D_o = external tube diameter, ft

L = tube length, ft.

The mass flow of the fluids is

$$M = \rho A_f \bar{V} \text{ lb/sec} \quad (5)$$

where ρ = fluid density, lb/ft^3

A_f = total cross section of fluid flow passage, ft^2

$\bar{V}$ = fluid velocity, ft/sec.

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[2] Δt_m is the logarithmic mean temperature difference between the primary and secondary fluids.

The over-all coefficient of heat transfer, U_o, can be obtained from the equation for circular cylinders,

$$\frac{1}{U_o} = \frac{D_o}{D_i h_i} + \frac{D_o \log_e(D_o/D_i)}{2k_m} + \frac{1}{h_o} \tag{6}$$

where h = film heat transfer coefficient, Btu/hr-ft^2-°F

k = thermal conductivity, Btu-ft/hr-ft^2-°F

and subscript m here refers to the metal tube.

As discussed in previous reports, it appears that the Lubarski-Kaufman equation is the most reliable one for the film coefficient for the liquid alkali metals. This equation is

$$h = 0.625\frac{k}{D}\left(\frac{DV\rho C}{k}\right)^{0.4}. \tag{7}$$

Adapting this equation to the shell side and using equations 1, 4, and 5 to eliminate the velocity,

$$h_o = 0.625\frac{k_o}{D_e}\left(\frac{4Q}{\pi k_o D_o N \Delta t_o}\right)^{0.4}, \tag{8}$$

where D_e is the equivalent outside diameter. It was shown in the first report (1) on this project that

$$D_f = \sqrt{D_e D_o} \tag{9}$$

where D_f is the outside flow diameter and equals $\sqrt{4A_f/\pi N}$. Equation 9 is based on the assumption that the tubes are placed on the apexes of equilateral triangles. In this first report (1), it was also shown that

$$D_e = D_o\left[\frac{2\sqrt{3}}{\pi}\left(\frac{\overline{P}}{D_o}\right)^2 - 1\right] \qquad (10)$$

where $\overline{P}$ is the pitch of the tubes.

At the mean temperature of the primary fluid of 1300°F, the thermal conductivity of sodium is 34.5 Btu/hr-ft^2-(°F/ft). Substituting this value into equation 8, the known value of Q, and the value of D_e from equation 10, and a pitch ratio of 1.272, equation 8 reduces to

$$h_o = 1584/D_o^{1.4}N^{0.4} \ . \qquad (11)$$

Using a value of k_m of 13.8 Btu/hr-ft^2-(°F/ft), the term

$$D_o\left[\log_e(D_o/D_i)\right]/2k_m = 0.006317D_o \, . \qquad (12)$$

The inside coefficient may be expressed as[3]

$$h_i = 0.023\frac{k_i}{D_i}\left(\frac{D_iG}{\mu_i}\right)^{0.8}\left(\frac{C_i\mu_i}{k_i}\right)^{0.4} . \qquad (13)$$

The mass rate of flow per unit area,

$$G = \frac{M}{A_f} = \frac{Q}{C_i\Delta t_iA_f} = \frac{3.413 \times 10^8}{C_i \times 500(0.84D_o)^2(\pi/4)N}$$

$$= \frac{1.232 \times 10^6}{C_iND_o^2} \text{ lb/sq ft/hr.}$$

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[3]This equation is for turbulent flow. Actually, for the 34-inch length tubes, the Reynolds number is slightly below that required for turbulent flow. By using the turbulent flow equation, the most favorable results are obtained; that is, h_i is overestimated. Even so, these indicate that the number of tubes required is prohibitively large. In subsequent calculations, the tube length is increased. For tube lengths of 35 inches and greater, the flow is turbulent.

Substituting this value into equation 13 and reducing,

$$h_i = \frac{1.775 \times 10^3 k_i^{0.6}}{D_o^{1.8}(C_i\mu_i)^{0.4}N^{0.8}} . \tag{14}$$

Substituting values from equations 11, 12, and 14 into equation 6,

$$\frac{1}{U_o} = \frac{D_o^{1.8}(C_i\mu_i)^{0.4}N^{0.8}}{0.84 \times 1.775 \times 10^3 k_i^{0.6}} + 0.006317D_o + \frac{D_o^{1.4}N^{0.4}}{1584}$$

or

$$\frac{1}{U_o} = \frac{D_o^{1.8}(C_i\mu_i)^{0.4}N^{0.8}}{1.491 \times 10^3 k_i^{0.6}} + 0.006317D_o + \frac{D_o^{1.4}N^{0.4}}{1584} . \tag{15}$$

Substituting this value of U_o into equation 2,

$$Q = \left[\frac{1}{\frac{D_o^{1.8}(C_i\mu_i)^{0.4}N^{0.8}}{1.491 \times 10^3 k_i^{0.6}} + 0.006317D_o + \frac{D_o^{1.4}N^{0.4}}{1584}}\right]\pi D_o LN(\Delta t)_m \tag{16}$$

The properties in equation 14 are to be evaluated at the mean inside film temperature. The mean inside stream temperature is 1250°F and the log mean temperature difference is 50°F. Hence, the mean inside film temperature is approximately 1265°F.

An examination of equation 16 shows that the gas requiring the smallest number of tubes will be that gas having the lowest value of $(C\mu)^{0.4}/k^{0.6}$. Of all the common gases, hydrogen has the lowest value for this term. Although hydrogen may not be as satisfactory as some other gases in certain respects, it has been selected for these caluclations because it will require the smallest secondary heat exchanger of all the common gases.

Following the procedure used in Report No. 1, it will be assumed, at first, that the secondary heat exchanger is located immediately adjacent to the heat source. The problem will be simplified by assuming that the tubes have a length of 34 inches and an outside diameter of 0.1 inch. The properties of hydrogen at the mean temperature of 1265°F and atmospheric pressure are as follows: C = 3.56 Btu/lb-°F; μ = 0.04739 lb/ft-hr; and k = 0.249 Btu/hr-ft^2-(°F/ft). It is contemplated that pressures will be investigated up to 4000 psia. Since hydrogen is at such an elevated temperature, even this pressure will not have a significant effect on these properties.[4] Substituting values into equation 16 and rearranging,

$$\frac{.008333N}{\frac{(3.56x.04739)^{0.4}N^{0.8}(.008333)^{1.8}}{1.491 \times 10^{3}(.249)^{0.6}} + .006317(.008333) + \frac{(.008333)^{1.4}N^{0.4}}{1584}}$$

$$= \frac{3.413 \times 10^{8} \times 12}{\pi \times 34 \times 50}.$$

$$\frac{0.008333N}{1.371x10^{-7}N^{0.8} + 5.264x10^{-5} + 7.75x10^{-7}N^{0.4}} = 7.669 \times 10^{5}. \quad (17)$$

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[4]At 1265°F, hydrogen has a reduced temperature of 28.8. At 4000 psia, the reduced pressure is 21.1. Referring to "The Properties of Gases and Liquids" (4, pp. 197, 238), by Reid and Sherwood, it may be seen that the pressure does not have a sufficient influence on the viscosity and the thermal conductivity of hydrogen to justify considering it. The effect of pressure on the specific heat of hydrogen may be seen to be very small by referring to "The Specific Heats of Certain Gases over Wide Ranges of Pressures and Temperatures" (5, p. 15).

Solving the above equation by trial and error,

$$N = 398{,}400 \text{ tubes.}$$

Pumping Power for Gas (Secondary Fluid)

The Darcy equation for pressure drop is

$$\Delta P = f\rho V^2 L/2g_c D \text{ lb/sq ft.} \tag{18}$$

For very smooth tubes,

$$f = 0.184/(Re)^{0.2}, \tag{19}$$

and

$$\Delta P = \frac{0.184 P\bar{V}^2 L}{(D\bar{V}\rho/\mu')^{0.2} 2g_c D} = \frac{0.184\rho^{0.8}\bar{V}^{1.8} L\mu'^{0.2}}{D^{1.2} 2g_c},$$

or

$$\Delta P = \frac{0.092\rho^{0.8}\bar{V}^{1.8} L\mu'^{0.2}}{D^{1.2} g_c}, \tag{20}$$

where μ' is the viscosity in lb/ft-sec and $\bar{V}$ is the velocity in ft/sec.

$$\bar{V} = \frac{M'}{\rho A_f} = \frac{94{,}800}{\rho C(500)(0.84D_o)^2 \frac{\pi}{4} N} = \frac{3422}{\rho C N D_o^2} \tag{21}$$

$$\bar{V}^{1.8} = \frac{3.644 \times 10^4}{(\rho CN)^{1.8} D_o^{3.6}} \tag{21a}$$

$$\Delta P = \frac{0.092\rho^{0.8} 3.644 \times 10^4 L\mu'^{0.2}}{(0.84D_o)^{1.2} g_c (\rho CN)^{1.8} D_o^{3.6}} = \frac{128.5 L\mu'^{0.2}}{\rho(CN)^{1.8} D_o^{4.8}}. \tag{22}$$

Substituting the properties of hydrogen into equation 22 and using a length of 34 inches and tube external diameter of 0.1 inch,

$$\Delta P = 0.02156/\rho \text{ lb/sq in.} \quad (23)$$

$$\text{Pumping power} = \frac{A_f \bar{V} \Delta P}{550}$$

$$= \frac{\pi (0.84 D_o)^2 N \times 342.2 \times 128.5 L \mu'^{0.2}}{4 \times 550 \rho C N D_o^2 \rho (CN)^{1.8} D_o^{4.8}}.$$

$$\text{Pumping power} = \frac{44.29 L \mu'^{0.2}}{\rho^2 C^{2.8} N^{1.8} D_o^{4.8}} \text{ hp.} \quad (24)$$

For hydrogen, and with a tube length of 34 inches and an external diameter of 0.1 inch, equation 24 becomes

$$\text{Pumping power} = 0.3006/\rho^2 \quad (25)$$

TABLE I

PERFORMANCE OF SECONDARY HEAT EXCHANGER
SECONDARY FLUID HYDROGEN
Tube length = 34 inches 398,400 tubes

Pressure psia	ρ lb_m/cu ft	$\bar{V}$ ft/sec	ΔP psi	Pumping horsepower
100	0.01090	318.8	1.973	2530
200	0.02180	159.4	0.9890	632.4
400	0.04360	79.71	0.4945	158.1
700	0.07631	45.55	0.2826	51.62
1000	0.1090	31.88	0.1973	25.30
2000	0.2180	15.94	0.0989	6.324
4000	0.4360	7.971	0.04945	1.581

In addition to the performance of the secondary fluid, the weight of the tubes in the heat exchanger and of the hydrogen in the tubes must be considered.

$$\text{Volume of metal in tubes} = (0.1^2 - 0.084^2)(\pi/4) \times 34$$
$$= 0.07862 \text{ cu in.}$$

Using a metal density of 0.29 lb/cu in,

$$\text{Weight per tube} = 0.29 \times 0.07862 = 0.02280 \text{ lb.}$$

For 398,400 tubes,

$$\text{Total weight of tubes} = 398{,}400 \times 0.02280$$
$$= 9{,}083 \text{ lb.}$$

$$\text{Internal volume per tube} = 0.084^2 \times (\pi/4) \times 34/1728$$
$$= 1.090 \times 10^{-4} \text{ cu ft.}$$

$$\text{Total internal volume} = 1.090 \times 10^{-4} \times 398{,}400$$
$$= 43.44 \text{ cu ft.}$$

At a hydrogen pressure of 4000 psia, the density of the hydrogen is 0.4360 lb/cu ft. The weight of hydrogen in the tubes at this pressure is 0.4360 x 43.44 = 18.94 lb. Thus it is evident that the weight of the hydrogen in the secondary heat exchanger may be neglected.

In making these calculations, the most favorable conditions were assumed. Hydrogen was selected as the gas to be investigated solely on the basis of its superior heat transfer characteristics. Very small tubes (0.1 in O.D.) were selected to obtain a large ratio of heat transfer area to volume. To minimize tube resistance, it was assumed that the tube wall was very thin (0.008 inch). This thickness was not increased, as it must be when very high gas pressures (i.e., 4000 psia) are used. In spite of all of these favorable assumptions, the number of tubes required is extremely large (398,400). This is to be compared with 14,700 tubes required when the secondary fluid is NaK 78. This means that the weight of the tubes when hydrogen is used will be approximately 27 times the weight of the tubes when the secondary fluid is NaK 78.

The pumping power required for the hydrogen is dependent on its density and hence on the pressure of the hydrogen. When the mean pressure is 100 psia, the pumping power requirements for the hydrogen is 2530 hp. If the mean hydrogen pressure is 1000 psia, the pumping power is 25.3 hp. This compares with a pumping power of 240 required for NaK 78 when it is used as the secondary fluid.

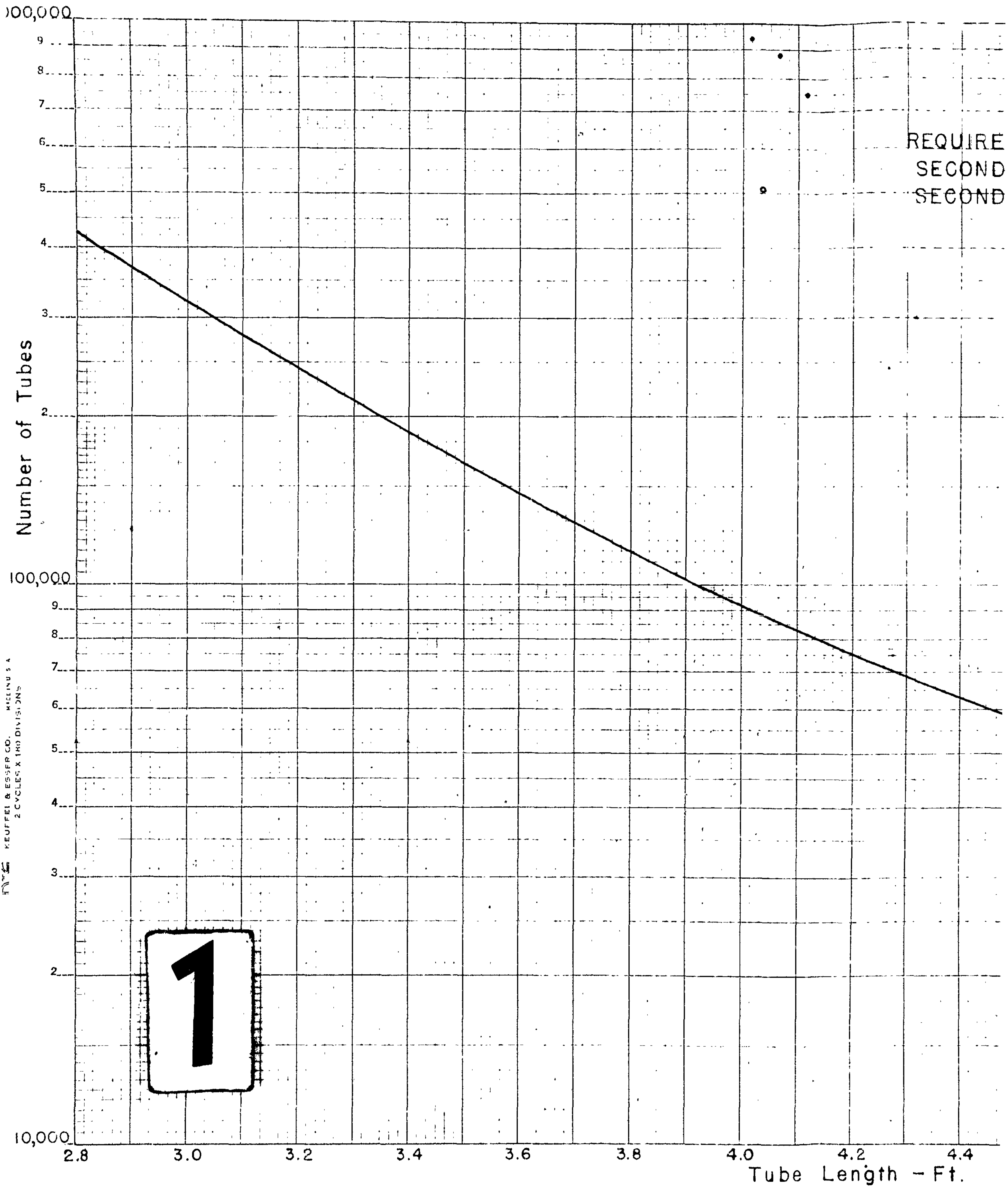
The size of the heat exchanger may be materially reduced by lengthening the tubes. Various tube lengths, ranging up to 72 inches, were substituted into equation 16 and the equation was solved by trial and error for the number of tubes, N, with the following results shown in Table II. Values for the 34-inch tube were repeated from Table I.

TABLE II

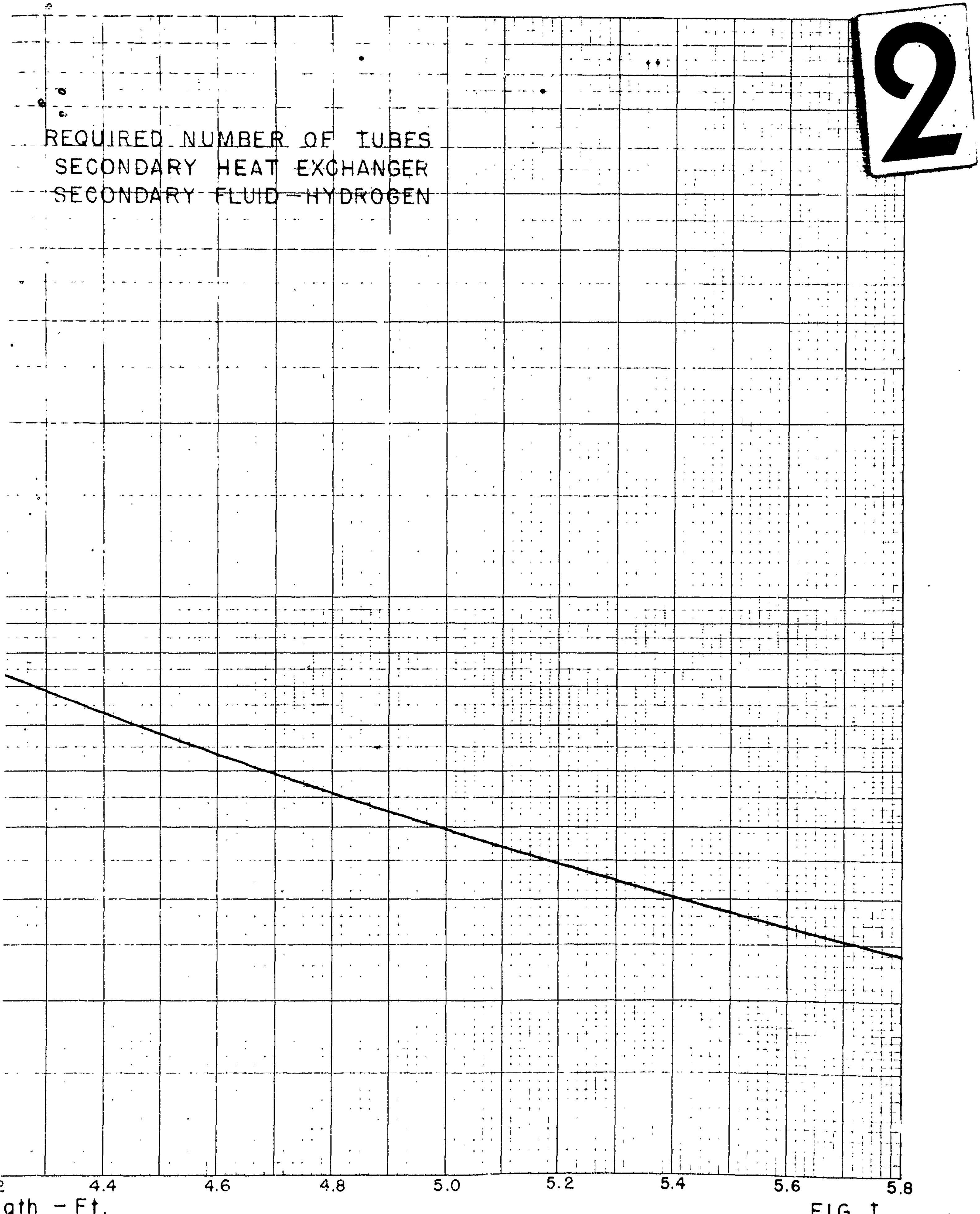
NUMBER OF TUBES IN SECONDARY HEAT EXCHANGER

Tube length in.	Number of tubes
34	398,400
48	91,300
60	39,600
68	26,020
72	21,550

Figure I is plotted from Table II.



REQUIRE
SECOND
SECOND
Number of Tubes
Tube Length – Ft.
100,000
10,000
2.8
3.0
3.2
3.4
3.6
3.8
4.0
4.2
4.4
KEUFFEL & ESSER CO.
2 CYCLES X 180 DIVISIONS
1



REQUIRED NUMBER OF TUBES
SECONDARY HEAT EXCHANGER
SECONDARY FLUID—HYDROGEN
4.4
4.6
4.8
5.0
5.2
5.4
5.6
5.8
gth - Ft.

FIG. I

Using the tube lengths from Table II, the fluid viscosities, pressure drops, and pumping powers may be calculated by use of equations 21, 22, and 24. The results of these calculations are shown in Tables III, IV, V, VI, VII, and VIII.

TABLE III

PERFORMANCE OF SECONDARY HEAT EXCHANGER
SECONDARY FLUID HYDROGEN
Tube length = 3.5 ft 168,000 tubes

Mean pressure psia	$\bar{V}$ ft/sec	ΔP psi	Pumping horsepower
400	194.8	3.056	975.8
700	111.3	1.748	318.6
1000	77.91	1.224	156.1
2000	38.95	0.6118	39.03
4000	19.48	0.3056	9.758

TABLE IV

PERFORMANCE OF SECONDARY HEAT EXCHANGER
SECONDARY FLUID HYDROGEN
Tube length = 4.0 ft 91,300 tubes

Mean pressure psia	$\bar{V}$ ft/sec	ΔP psi	Pumping horsepower
400	347.5	9.900	3166
700	198.7	5.675	1034
1000	139.0	3.960	506.5
2000	69.50	1.980	126.6
4000	34.75	0.990	31.66

TABLE V

PERFORMANCE OF SECONDARY HEAT EXCHANGER
SECONDARY FLUID HYDROGEN
Tube length = 4.5 ft 57,400 tubes

Mean pressure psia	$\bar{V}$ ft/sec	ΔP psi	Pumping horsepower
400	553.1	25.68	8211
700	316.1	14.67	2681
1000	221.2	10.27	1314
2000	110.6	5.136	328.5
4000	55.31	2.568	82.11

TABLE VI

PERFORMANCE OF SECONDARY HEAT EXCHANGER
SECONDARY FLUID HYDROGEN
Tube length = 5.0 ft 39,600 tubes

Mean pressure psia	$\bar{V}$ ft/sec	ΔP psi	Pumping horsepower
400	801.2	55.66	17,800
700	457.8	31.81	5,811
1000	320.5	22.26	2,848
2000	160.2	11.13	711.9
4000	80.12	5.566	178.0

TABLE VII

PERFORMANCE OF SECONDARY HEAT EXCHANGER
SECONDARY FLUID HYDROGEN
Tube length - 5.5 ft 28,700 tubes

Mean pressure psia	$\bar{V}$ ft/sec	ΔP psi	Pumping horsepower
400	1106	109.3	34,950
700	632.1	62.46	11,410
1000	442.5	43.72	5,592
2000	221.2	21.86	1,398
4000	110.6	10.93	349.5

TABLE VIII

PERFORMANCE OF SECONDARY HEAT EXCHANGER
SECONDARY FLUID HYDROGEN

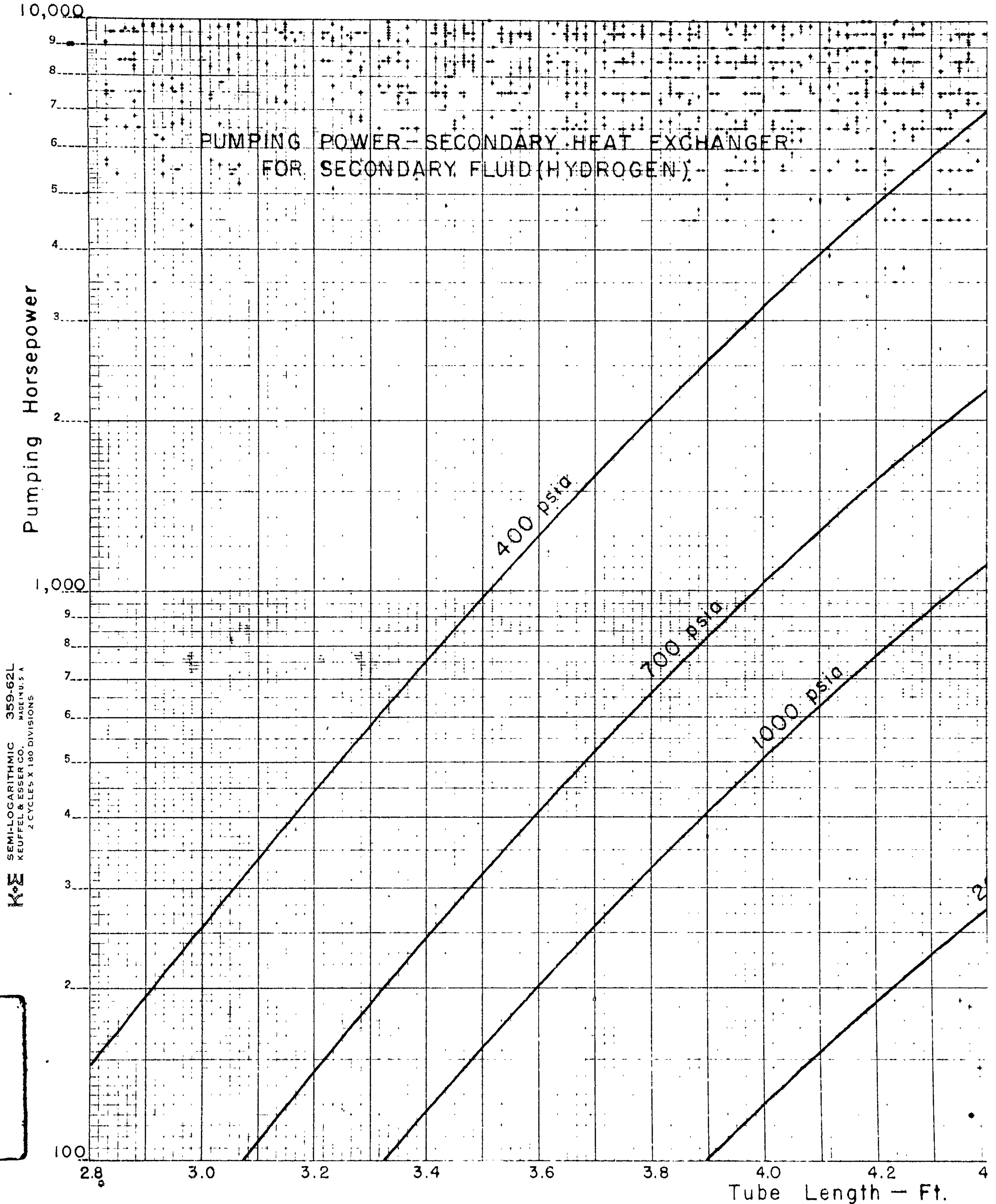
Tube length = 5.8 ft 24,100 tubes

Mean pressure psia	$\overline{V}$ ft/sec	ΔP psi	Pumping horsepower
400	1317	157.8	50,590
700	752.7	90.20	16,480
1000	526.9	63.14	8,075
2000	263.5	31.57	2,019
4000	131.7	15.78	505.9

Figure II shows the relationship between pumping power requirements and the tube length for various gas pressures. This figure is based on the data shown in Tables I, III, IV, V, VI, VII, and VIII.

It was shown previously that the weight of the hydrogen contained in the tubes of the secondary heat exchanger was only 19 pounds when the number of tubes was 398,400 and the gas pressure was 4000 psia. Thus, in general, the weight of the hydrogen in the tubes may be neglected for the secondary heat exchanger. On the other hand, when the secondary fluid is NaK 78 and a tube length of 34 inches is used, there are over 100 pounds of secondary fluid in the tubes.

In Report No. 1 the following equation was developed for determining the pumping power required for the primary fluid in the secondary heat exchanger:


PUMPING POWER - SECONDARY HEAT EXCHANGER
FOR SECONDARY FLUID (HYDROGEN)
Pumping Horsepower
Tube Length — Ft.
400 psia
700 psia
1000 psia
10,000
1,000
100
2.8
3.0
3.2
3.4
3.6
3.8
4.0
4.2
SEMI-LOGARITHMIC 359-62L
KEUFFEL & ESSER CO. MADE IN U.S.A.
2 CYCLES X 180 DIVISIONS

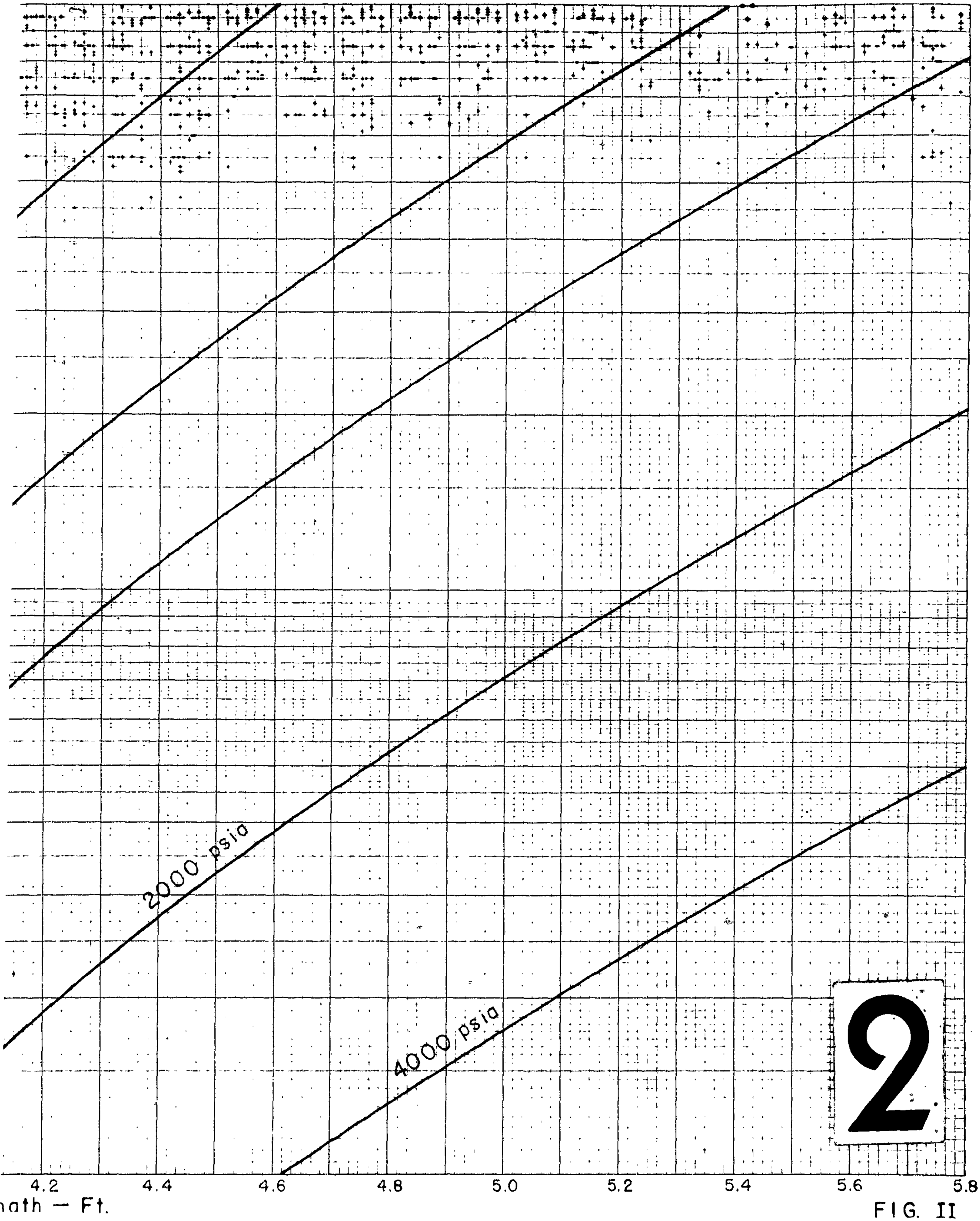


FIG. II

$$(p \cdot p \cdot)_o = \frac{0.01527 L'' \mu_o^{0.2}}{\left[1.1027(\overline{P}/D_o)^2 - 1\right]^3 D_o''^{4.8} \overline{N}^{1.8} C_o^{2.8} \rho_o^2} \qquad (26)$$

where p·p· = pumping horsepower

L" = tube length, in.

D" = tube diameter, in.

$\overline{N}$ = number of tubes x 10^{-4}

and subscript o refers to the primary fluid (on the outside of the tubes).

TABLE IX

PUMPING POWER FOR PRIMARY FLUID
SECONDARY FLUID HYDROGEN

Tube length in.	Pumping power hp
34	0.28
48	5.79
60	31.6
72	116.8

These figures are to be compared with a value of 69 hp for the primary fluid when the secondary fluid is NaK 78.

These calculations show that the number of tubes required decreases very rapidly as the tube length is increased. For example, when the tube length is increased from 34 inches to 60 inches, the number of tubes is reduced from 398,400 to 39,600. A further increase in the tube length will reduce the number of tubes required to a value equal to that required for the liquid alkali metals. However, there is a rapid increase in the pumping power requirements with a decrease in the number of tubes. With tubes having a length of 60 inches, the power requirements are 2,844 hp with a hydrogen pressure of 1,000 psia and 178 hp at 4,000 psia.

It should be evident that the size and weight of the secondary heat exchanger for hydrogen as the secondary fluid can be made comparable with those using a liquid alkali only by using several times as much pumping horsepower, and only then by using very high gas pressures. The pumping power for the hydrogen is fantastic for low values of pressure. As an example, for a hydrogen pressure of 400 psia and a tube length of 5.8 feet, the pumping power is 50,590 horsepower.

In summary, when short tubes are used (34 inches in this case), the horsepower requirement for hydrogen can be made much less than that required for the liquid alkali metals (except lithium) by using gas pressures of approximately 700 to 1,000 psia. However, the number of tubes (and hence exchanger weight and volume) required for hydrogen may be 25 to 30 times that for the liquid alkali metals. When longer tubes are used, there is a very sharp reduction in the number of tubes required, but there is a corresponding increase in the horsepower requirement. Unless high pressures (i.e., 4,000 psia) are used, the pumping power requirements for hydrogen in long tubes (i.e., 7 feet) are set at least several times those for the liquid alkali metals in 34-inch tubes.

III. TRANSFER PIPING SYSTEM

In considering the transfer piping system, three factors are of prime consideration: (1) the weight of the pipeline, (2) the weight of the fluid contained in the pipeline, and (3) the power required to force the secondary fluid through the pipeline. When the distance between the heat source and the air radiators is large, the problem of maintaining low total weight and low pumping power is a very real one.

It is necessary to use an extremely high grade piping material in order that high pressure may be used without excessive pipeline weight. (Pumping power requirements drop very rapidly with an increase in gas pressure.) Certain alloys of the cast nature whose ultimate strength is at least 110,000 psi at 1500°F are now available. It does not seem too unreasonable to expect that alloys having this value of ultimate strength and which will be satisfactory for containing a heat transfer gas will be developed. Assuming that such materials will be available, it seems reasonable that schedule No. 40 pipes will stand pressures up to 1,000 psia, No. 80 up to 2,000 psia, and No. 160 up to 4,000 psia. Further refinements in manufacture may allow much larger pressures to be used in a given weight pipe.

Since there are two air radiators, it will be assumed that the energy transferred to each radiator is 50 megawatts, or 47,400 Btu per second. Calculations will be made for the transfer piping for one air radiator. The length of the transfer piping is a function of the configuration of the propulsion system. In the calculations which follow, determinations were made for an equivalent length of the piping system of 10 feet. Since the mean temperature in the return or cold leg is materially different from that of the hot leg, separate calculations are made for the two legs.

Neglecting heat losses, the mean temperature is 1500°F in the hot leg and 1000°F in the cold leg. The mass rate of flow, $M' = Q/C\Delta t = 47{,}400/3.56 \times 500 = 26.63$ lb/sec. For the hot leg, $\rho = p/RT = 144P/(1545/2)(1960) = 9.510 \times 10^{-5}P$ lb_m/cu ft, where P is in lb/sq in. abs.

The Darcy equation for pressure drop is

$$\Delta P = f\rho\overline{V}^2L/2g_cD \text{ lb/sq ft.} \tag{27}$$

Assuming that the pipes may be treated as very smooth tubes,

$$f = 0.184/(Re)^{0.2}$$

$$\Delta P = \frac{0.184\rho\overline{V}^2L}{2g_c(D\overline{V}\rho/\mu')^{0.2}D} = \frac{0.184\rho^{0.8}\overline{V}^{1.8}L(\mu')^{0.2}}{2g_cD^{1.2}}$$

$$\Delta P = \frac{2.860 \times 10^{-3}\rho^{0.8}\overline{V}^{1.8}L(\mu')^{0.2}}{D^{1.2}} \text{ lb/sq ft.} \tag{28}$$

But

$$\overline{V} = M'/\rho A_f = 26.63/\rho(\pi/4)D^2 = 33.91/\rho D^2, \tag{29}$$

or

$$\overline{V}^{1.8} = 568.2/\rho^{1.8}D^{3.6}.$$

Substituting into equation 28,

$$\Delta P = \frac{2.860 \times 10^{-3}\rho^{0.8} \times 568.2L(\mu')^{0.2}}{D^{1.2}\rho^{1.8}D^{3.6} \times 144}$$

$$\Delta P = \frac{1.129 \times 10^{-2}L(\mu')^{0.2}}{\rho D^{4.8}} \text{ lb/sq in.} \qquad (30)$$

$$\text{Pumping power} = A\bar{V}\Delta P/550$$

$$= \frac{\pi D^2 \times 33.91 \times 1.129 \times 10^{-2}L(\mu')^{0.2} \times 144}{4 \times \rho D^2 \times \rho D^{4.8} \times 550}.$$

$$\text{Pumping power} = \frac{0.07868L(\mu')^{0.2}}{\rho^2 D^{4.8}}. \qquad (31)$$

Hot Leg Calculations

For the hot leg at a mean hydrogen temperature of 1500°F, $\mu' = 14.31 \times 10^{-6}$ lb/sec-ft. Using a value of L of 10 feet and substituting into equation 30,

$$\Delta P = 0.01212/\rho D^{4.8} \text{ lb/sq in.} \qquad (32)$$

Likewise, equation 31 reduces to

$$\text{Pumping power} = 0.08453/\rho^2 D^{4.8}. \qquad (33)$$

TABLE X

TRANSFER PIPE PERFORMANCE
HOT LEG--LENGTH 10 FEET
SECONDARY FLUID HYDROGEN

	Schedule 40 pipe		Gas pressure = 1,000 psia		
Nominal diam., in.	8	10	12	14	16
I.D., in.	7.981	10.02	11.94	13.13	15.00
I.D., ft	0.6651	0.8350	0.9950	1.094	1.250
$D_i^{4.8}$	0.1412	0.4208	0.9762	1.540	2.919
Pressure drop, psi	0.9029	0.3029	0.1306	0.08277	0.4368
Pumping hp	66.19	22.21	8.573	6.068	3.202
	Schedule 80 pipe		Gas pressure = 2,000 psia		
Nominal diam., in.	8	10	12	14	16
I.D., in.	7.625	9.56	11.38	12.50	14.31
I.D., ft	0.6354	0.7967	0.9483	1.042	1.192
$D_i^{4.8}$	0.1134	0.3358	0.7752	1.216	2.328
Pressure drop, psi	0.5620	0.1898	0.0822	0.0524	0.0274
Pumping hp	20.60	6.957	3.014	1.921	1.004
	Schedule 160 pipe		Gas pressure = 4,000 psia		
Nominal diam., in.	8	10	12	14	16
I.D., in.	6.813	8.500	10.13	11.19	12.88
I.D., ft	0.5697	0.7083	0.8442	0.9325	1.073
$D_i^{4.8}$	0.06606	0.1911	0.4435	0.7150	1.404
Pressure drop, psi	0.4834	0.1668	0.0719	0.0446	0.0227
Pumping hp	8.836	3.057	1.317	0.817	0.416

Cold Leg Calculations

For the cold leg at a mean hydrogen temperature of 1000°F, $\mu' = 11.79 \times 10^{-6}$ lb/sec-ft. Using a value of L of 10 feet and substituting into equation 30,

$$\Delta P = 0.1166/\rho D^{4.8} \text{ lb/sq in.} \qquad (34)$$

Likewise, equation 31 reduces to

$$\text{Pumping power} = 0.08132/\rho^2 D^{4.8}. \qquad (35)$$

The performance characteristics of the cold leg are shown in Table XI.

Hot and Cold Legs Combined

It has been assumed that the strength of the piping is not sufficiently higher at 1000°F than at 1500°F to permit using any higher gas pressure in the cold leg than in the hot leg for any given weight pipe. (Actually, because of added strength at the lower temperature, allowable gas pressures may be increased by 10 to 15 percent.) Since the purpose of this report is to investigate the possibilities of using a gaseous secondary fluid rather than to establish a precise relationship between the performance of a gaseous fluid and the liquid alkali metals, the complications, caused by increasing the gas pressure in the cold leg, cannot be justified in this report. Hence a uniform gas pressure will be assumed for the two legs. The weight of the transfer piping for a length of 10 feet is shown in Table XII and is plotted in Figure III.

TABLE XI

TRANSFER PIPE PERFORMANCE
COLD LEG--LENGTH 10 FEET
SECONDARY FLUID HYDROGEN

	Schedule 40 pipe		Gas pressure = 1,000 psia		
Nominal diam., in.	8	10	12	14	16
I.D., in.	7.981	10.02	11.94	13.13	15.00
I.D., ft	0.6651	0.8350	0.9950	1.094	1.250
$D_i^{4.8}$	0.1412	0.4208	0.9762	1.540	2.919
Pressure drop, psi	0.6470	0.2170	0.0936	0.0593	0.0313
Pumping hp	35.33	11.85	5.110	3.239	1.709
	Schedule 80 pipe		**Gas pressure = 2,000 psia**		
Nominal diam., in.	8	10	12	14	16
I.D., in.	7.625	9.56	11.38	12.50	14.31
I.D., ft	0.6354	0.7967	0.9483	1.042	1.192
$D_i^{4.8}$	0.1134	0.3358	0.7752	1.216	2.328
Pressure drop, psi	0.4027	0.1360	0.0589	0.0376	0.0196
Pumping hp	11.00	3.713	1.609	1.025	0.536
	Schedule 160 pipe		**Gas pressure = 4,000 psia**		
Nominal diam., in.	8	10	12	14	16
I.D., in.	6.813	8.500	10.13	11.19	12.88
I.D., ft	0.5697	0.7083	0.8442	0.9325	1.073
$D_i^{4.8}$	0.06606	0.1911	0.4435	0.7150	1.404
Pressure drop, psi	0.3457	0.1195	0.0515	0.0319	0.0163
Pumping hp	4.719	1.632	0.703	0.436	0.222

TABLE XII

TRANSFER PIPING
WEIGHT PER 10-FOOT LENGTH
SECONDARY FLUID HYDROGEN

Nominal diam. in.	O.D. in.	$\frac{\pi D_o^2}{4}$	I.D. in.	$\frac{\pi D_i^2}{4}$	Wall area sq in.	Metal volume cu in.	Weight of pipe lb
			Schedule 40				
8	8.625	58.42	7.981	50.03	8.37	1004	291.3
10	10.75	90.76	10.02	78.85	11.91	1429	414.5
12	12.75	127.67	11.94	111.97	15.70	1884	546.2
14	14.0	153.93	13.13	135.40	18.53	2224	644.8
16	16.0	201.04	15.00	176.71	24.33	2919	846.7
			Schedule 80				
8	8.625	58.42	7.625	45.66	12.76	1531	444.0
10	10.75	90.76	9.56	71.78	18.98	2278	660.5
12	12.75	127.67	11.38	101.70	25.96	3115	903.4
14	14.0	153.93	12.50	122.72	31.21	3745	1086
16	16.0	201.04	14.31	160.83	40.21	4825	1399
			Schedule 160				
4	4.500	15.90	3.438	9.283	6.717	806.0	233.8
6	6.625	34.47	5.189	21.14	13.23	1588	460.4
8	8.625	58.42	6.813	36.45	21.97	2636	764.6
10	10.75	90.76	8.500	56.74	34.02	4082	1184
12	12.75	127.67	10.13	80.59	47.08	5650	1638
14	14.0	153.93	11.19	98.34	55.59	6671	1935
16	16.0	201.04	12.88	130.29	70.75	8490	2462

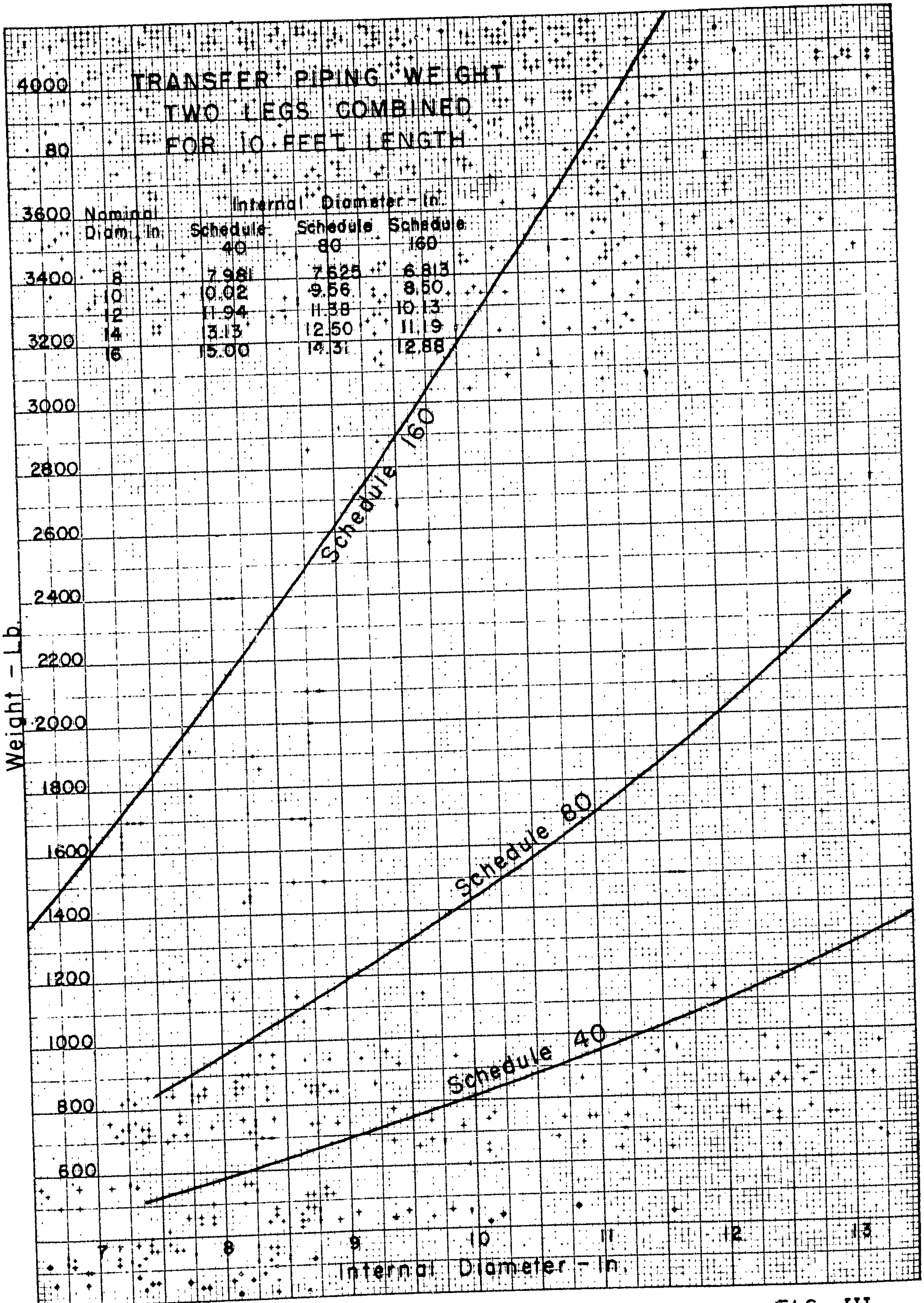

TRANSFER PIPING WEIGHT
TWO LEGS COMBINED
FOR 10 FEET LENGTH
Internal Diameter - In.
Nominal Diam., In.
Schedule 40
Schedule 80
Schedule 160
8 7.981 7.625 6.813
10 10.02 9.56 8.50
12 11.94 11.38 10.13
14 13.13 12.50 11.19
16 15.00 14.31 12.88
Weight - Lb.
4000
3800
3600
3400
3200
3000
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
7
8
9
10
11
12
13
Internal Diameter - In.
Schedule 160
Schedule 80
Schedule 40

FIG. III

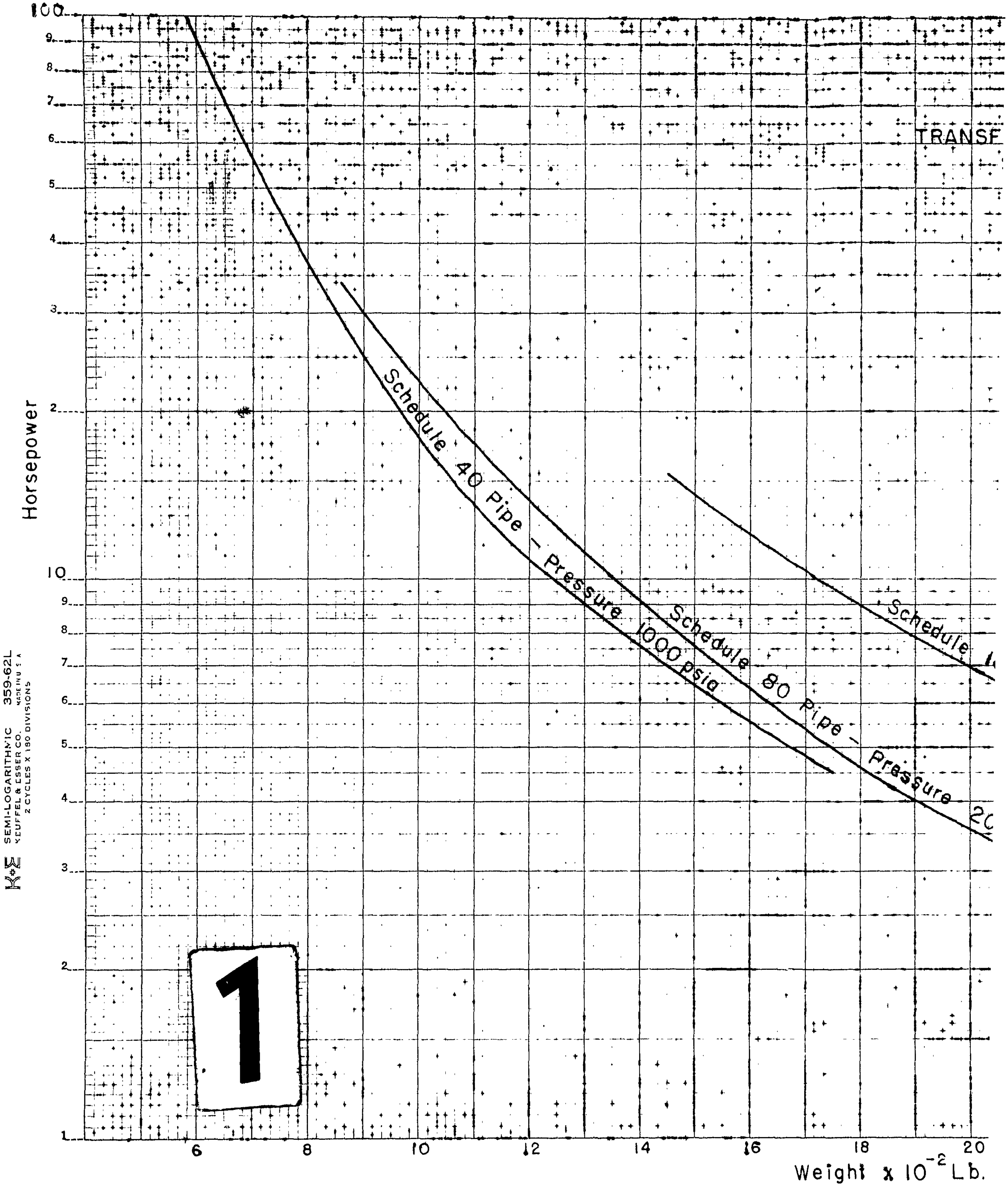
The combined weight of the two legs and the combined pumping power required for a length of 10 feet are shown in Table XIII.

TABLE XIII

COMBINED PERFORMANCE OF TRANSFER PIPING
TWO LEGS--10-FOOT LENGTH
SECONDARY FLUID HYDROGEN

	Piping weight, lb				
Nominal diam., in.	8	10	12	14	16
Schedule 40	582.6	829.0	1092.4	1289.4	1693.4
Schedule 80	888.0	1321.0	1806.8	2172	2798
Schedule 160	1529.2	2368	3272	3870	4924
	Pumping power, hp				
Nominal diam., in.	8	10	12	14	16
Schedule 40	101.52	34.06	13.68	9.31	4.91
Schedule 80	31.60	10.67	4.62	2.95	1.54
Schedule 160	13.56	4.69	2.02	1.25	0.64

Figure IV shows a relationship between the pumping horsepower requirements and the transfer piping weight for the two legs combined, for a piping length of 10 feet. Inspection of this figure shows that the weight-pumping requirements are very much higher when a schedule 160 pipe is used, and that the schedule 40 pipe is slightly superior in this respect to the schedule 80 pipe. Hence it will be assumed that a schedule 40 pipe will be used with gas pressures of 1000 psia.



100
9
8
7
6
5
4
3
2
10
9
8
7
6
5
4
3
2
1
6
8
10
12
14
16
18
20
Horsepower
Weight x 10^{-2} Lb.
TRANSF
Schedule 40 Pipe - Pressure 1000 psia
Schedule 80 Pipe - Pressure 20
Schedule
SEMI-LOGARITHMIC 359-62L
KEUFFEL & ESSER CO. MADE IN U.S.A.
2 CYCLES X 180 DIVISIONS
1

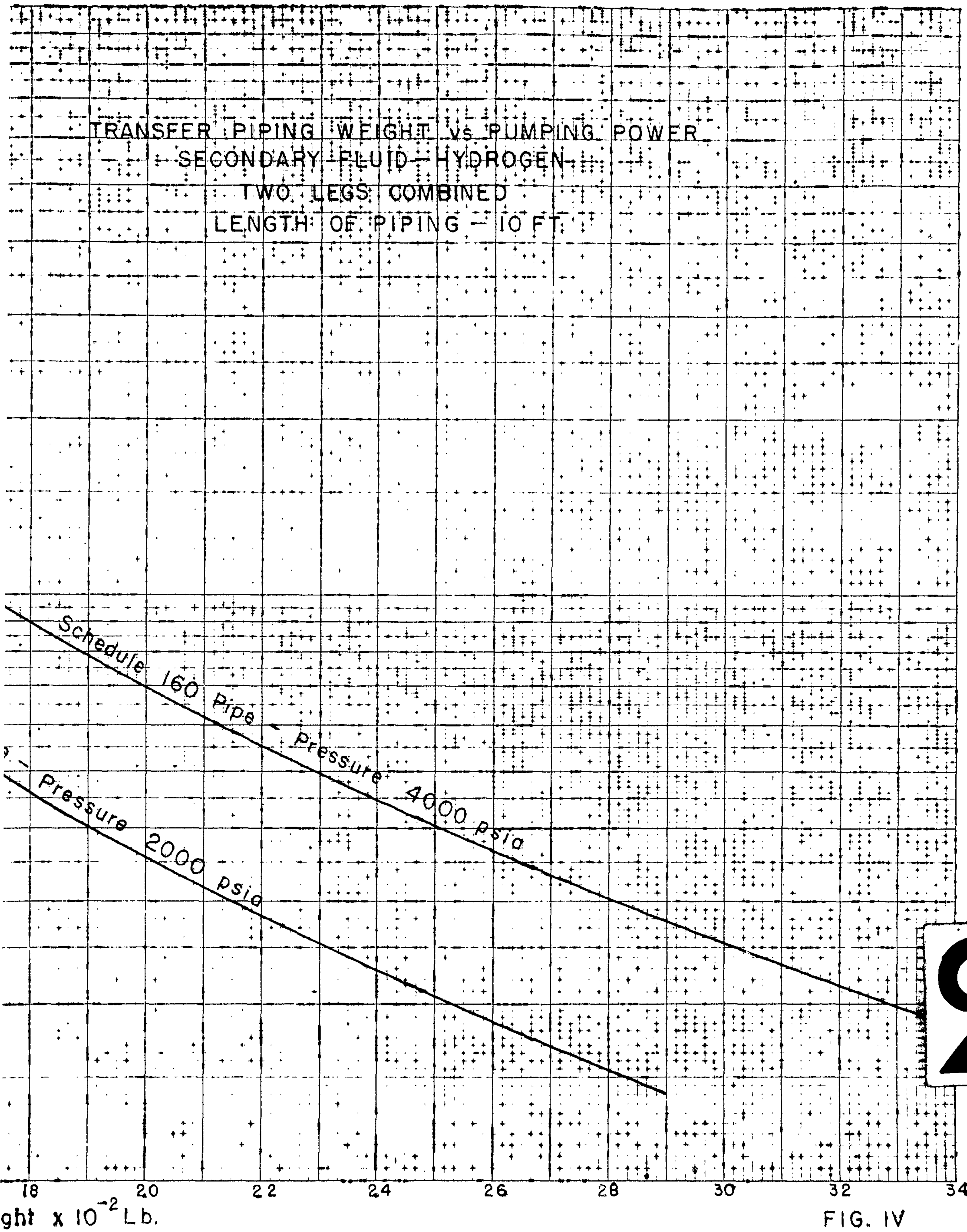

TRANSFER PIPING WEIGHT vs PUMPING POWER
SECONDARY FLUID — HYDROGEN
TWO LEGS COMBINED
LENGTH OF PIPING — 10 FT
Schedule 160 Pipe - Pressure 4000 psia
Pressure 2000 psia
18
20
22
24
26
28
30
32
34
ght x 10^{-2} Lb.

FIG. IV

The pumping horsepower requirements as a function of the internal pipe diameter are shown for various weight pipes in Figure V.

For the largest pipe and the highest pressures considered here, the weight of the hydrogen present in the two legs of transfer piping is approximately 8 pounds for the 10 feet of the piping system. Hence the weight of the hydrogen may be neglected and the weight of the piping system and its contained fluid taken as the weight of the pipe itself.

A comparison between the transfer piping for hydrogen with that for NaK 78 shows that, for a fixed pumping power, the weight of the pipeline and its contained fluid for hydrogen is roughly one and three quarters times that for NaK 78. A comparison between hydrogen and a heavy liquid metal, such as bismuth, shows that, for a given pumping power, the weight of the piping system and its contained fluid for bismuth is roughly one and three quarters times that for hydrogen.

On the other hand, for a fixed weight of the transfer piping and its contained fluid, the pumping requirements for hydrogen are roughly five times those for NaK 78. Likewise, for a fixed total weight of the piping system, the pumping power required for bismuth is roughly five times that for hydrogen.

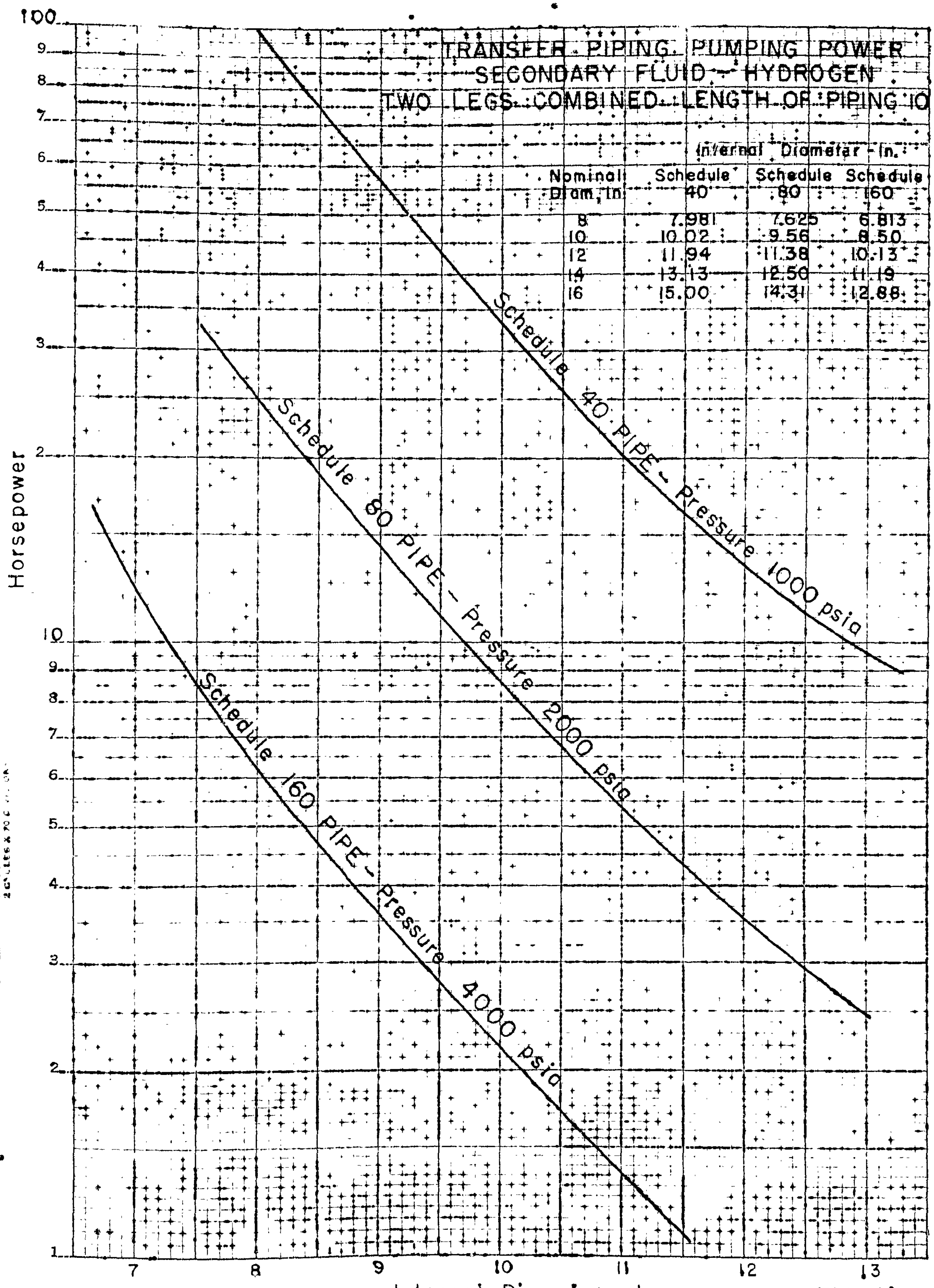

TRANSFER PIPING PUMPING POWER
SECONDARY FLUID - HYDROGEN
TWO LEGS COMBINED LENGTH OF PIPING 10
Internal Diameter - In.
Nominal Diam, In | Schedule 40 | Schedule 80 | Schedule 160
8 | 7.981 | 7.625 | 6.813
10 | 10.02 | 9.56 | 8.50
12 | 11.94 | 11.38 | 10.13
14 | 13.13 | 12.50 | 11.19
16 | 15.00 | 14.31 | 12.88
Schedule 40 PIPE - Pressure 1000 psia
Schedule 80 PIPE - Pressure 2000 psia
Schedule 160 PIPE - Pressure 4000 psia
Horsepower
100
9
8
7
6
5
4
3
2
10
9
8
7
6
5
4
3
2
1
7
8
9
10
11
12
13
Internal Diameter -- In.

FIG. V

IV. AIR RADIATOR

There is a very large number of possible configurations for the air radiator. In our report (6) entitled, "The Effect of Secondary Heat Transfer Fluids on Air Radiator Characteristics," two air radiators, having widely different geometries, were selected. This report showed that approximately the same relative comparisons for the various liquid alkali metals existed for the two types of air radiators. Since it is the purpose of this report to investigate whether or not gases deserve further consideration as secondary heat transfer fluids and not to establish a comparative merit number for them, only one type of air radiator will be considered here. Because of its simplicity and the ease of its analysis, the plain tube air radiator will be used, even though it is not a very practical air radiator for this type of service.

The following assumptions are made relative to the air radiator:

1. The tubes are stainless steel, with an external diameter of 0.1 inch, an internal diameter of 0.084 inch, a length of 7 feet, and a thermal conductivity of 13.8 Btu/hr-ft^2(°F/ft).

2. Rate of heat transfer = 47,400 Btu/sec (50 megawatts).

3. The secondary fluid (hydrogen) cools from 1500°F to 1000°F.

4. The air is heated from 400°F to 1350°F.

5. The air flows across plain tubes, arranged in banks, and the secondary fluid flows through the tubes.

6. The air pressure at radiator entrance is 30 psia.

7. There is a pressure drop of 3 psia (10 percent) of the air in the air radiator.

8. Both the longitudinal and the transverse pitch of the tubes will be taken as $1.5D_o$, where D_o is the external diameter of the tubes.

McAdams (7, p. 163), in his "Heat Transmission," presents the following equation for air pressure drop across a bank of tubes.

$$P_1 - P_2 = 4f''R(G'_{max})^2/2g\rho \text{ lb/sq ft} \tag{36}$$

where f'' = the friction factor

R = number of rows of tubes in the direction of flow

G'_{max} = lb/sec ÷ min flow area, sq ft

ρ = mean density.

McAdams (7) also presents the following equation for the friction factor.

$$f'' = \left[0.23 + \frac{0.11}{(X_T-1)^{1.08}}\right]\left(\frac{D_o G_{max}}{\mu_f}\right)^{-0.15} \tag{37}$$

where X_T is the transverse pitch ratio (assumed to be 1.5 here), and μ_f is the viscosity at the mean film conditions. The mass rate of flow of air is 47,400/(452.44-206.46) = 192.7 lb/sec. The minimum free flow area in between the tubes is [illegible] $(1.5-1.0)D_oLN/R$, where D_o is the external tube diameter, L is the length of the tubes, and N is the number of tubes. Then,

$$G_{max} = \frac{192.7 \times 3600 \times R}{0.5D_oLN} = 1.3874 \times 10^6 \frac{R}{D_oLN} \text{ lb/sq ft-hr}, \tag{38}$$

and

$$G'_{max} = 385.4R/D_oLN \text{ lb/sq ft-sec.} \tag{38a}$$

It is necessary to establish the mean air film temperature to evaluate the viscosity, μ, in equation 37. Assuming pure counterflow heat transfer,

$$\text{log mean } \Delta t = \frac{(1000-400) - (1500-1350)}{\log_e [(1000-400)/(1500-1350)]} = 324.6F.$$

For the conditions of this problem, Bowman, Mueller, and Nagle (8), in their paper, "Mean Temperature Differences in Design," give a correction factor of 0.55 to be multiplied by the log mean Δt for complete mixing of the air in the heat exchanger and 0.76 for no mixing. Since there will be some baffles present, a correction factor of 0.6 will be used here as being sufficiently accurate for the comparison purposes of this report. Then the true mean temperature difference is $0.6 \times 324.6 = 194.7°F$. The approximate mean air film temperature is $(1500+1000)/2 - 194.7/2 = 1153°F$. This mean air film temperature is based on the assumption that all of the temperature drop occurs in the air film. Although this statement is not accurate, this temperature is sufficiently close to the true value to permit a reasonably accurate determination of the air film properties to satisfy the purpose of this report. Substituting into equation 37,

$$f'' = \left(0.23 + \frac{0.11}{0.473}\right)\left(\frac{D_o \times 1.3874 \times 10^6 R}{D_o LN \times 0.095}\right)^{-0.15}$$

$$f'' = 0.03895(LN/R)^{0.15}. \tag{39}$$

Substituting into equation 36,

$$P_1 - P_2 = \frac{4R(0.03895)(LN/R)^{0.15}}{2g\rho_m}(385.4R/D_o LN)^2$$

$$= \frac{359.7R^{2.85}}{(LN)^{1.85}D_o{}^2\rho_m} \text{ lb/sq ft},$$

or

$$P_1 - P_2 = \frac{2.498R^{2.85}}{(LN)^{1.85}D_o{}^2\rho_m} \text{ lb/sq in.} \tag{40}$$

Substituting a value of 3.0 psi for P_1-P_2, a value of 7.0 feet for L, and a value of 0.1/12 for D_o, equation 40 becomes

$$3 = \frac{2.498R^{2.85}}{7^{1.85}N^{1.85}(0.1/12)^2\rho_m}$$

or

$$N = \left[\frac{2.498R^{2.85}}{3 \times 7^{1.85}(0.1/12)^2\rho_m}\right]^{1/1.85}$$

$$N = 22.89R^{1.541}/\rho_m{}^{1/1.85}. \tag{41}$$

At the air stream mean temperature of 1055°F and a mean pressure of 28.5 psia, the mean air density is 0.05078 lb/cu ft. Substituting this density into equation 41,

$$N = 114.7R^{1.541}. \tag{42}$$

Since equation 42 involves two unknowns, it is necessary to establish a second equation containing these unknowns.

$$Q = U_o A_o (\Delta t)_m$$
$$= 47{,}400 \times 3{,}600 = 1.706 \times 10^8 \text{ Btu/hr}, \tag{43}$$

and

$$U_o = \frac{1}{\frac{1}{h} + \frac{D}{2k_m} \log_e \frac{D_o}{D_i} + \frac{D_o}{D_i h_i}}. \tag{44}$$

For this type of heat exchanger and for the given conditions, McAdams (7) recommends the following equation for h_o:

$$h_o D_o / k_f = 0.48 (D_o G_{max} / \mu_f)^{0.562}. \tag{45}$$

Although equation 45 is valid only for a heat exchanger having ten rows of tubes, the variation in h_o caused by using the number of rows anticipated here is not sufficient to justify modifying equation 45 as far as this report is concerned. Then

$$h_o = \frac{0.46 \times 0.0393}{0.1/12} \left[\frac{(0.1/12) \times 1.3874 \times 10^6 R}{0.095 (0.1/12) 7N} \right]^{0.562}$$

$$h_o = 7743 (R/N)^{0.562}. \tag{46}$$

$$\frac{D_o}{2k_m} \log_e \frac{D_o}{D_i} = \frac{(0.1/12)}{2 \times 13.8} \log_e \frac{1}{0.84} = 0.0000527. \tag{47}$$

Equation 13 is valid for the inside film coefficient for each air radiator,

$$G = \frac{M}{A_f} = \frac{Q}{C_i \Delta t_i A_f} = \frac{3.413 \times 10^3 \times 5.0 \times 10^4}{C_i \times 500 D_i^2 \times (\pi/4) N}$$

$$= \frac{4.346 \times 10^5}{C_i D_i^2 N} \text{ lb/sq ft/hr.}$$

Substituting into equation 13,

$$h_i = 0.023 \frac{k_i}{D_i} \left(\frac{D_i \times 4.346 \times 10^5}{N C_i D_i^2 \mu_i} \right)^{0.8} (C_i \mu_i / k_i)^{0.4}$$

$$= 745.0 \left(\frac{k_i^{0.6}}{(C_i \mu_i)^{0.4} D_i^{1.8} N^{0.8}} \right).$$

$$h_i = 1.020 \times 10^3 \left(\frac{k_i^{0.6}}{(C_i \mu_i)^{0.4} D_o^{1.8} N^{0.8}} \right). \quad (48)$$

The mean properties of hydrogen will not differ a significant amount from those used in the secondary heat exchanger. Substituting the values of these properties into equation 48,

$$h_i = 1.020 \times 10^3 \left(\frac{0.249^{0.6}}{(3.56 \times 0.04739)^{0.4} 0.008333^{1.8} N^{0.8}} \right)$$

$$h_i = 4.999 \times 10^6 / N^{0.8}. \quad (49)$$

Substituting equations 46, 47, and 49 into equation 44,

$$\frac{1}{U_o} = (1/7743)(N/R)^{0.562} + 0.0000527 + \left[N^{0.8} / (0.84 \times 4.996 \times 10^6) \right]$$

$$\frac{1}{U_o} = 1.291 \times 10^{-4} (N/R)^{0.562} + 5.27 \times 10^{-5} + 2.382 \times 10^{-7} N^{0.8}. \quad (50)$$

Substituting equation 50 into equation 43,

$$\left[\frac{1}{1.291\times10^{-4}\left(\frac{N}{R}\right)^{0.562}+5.27\times10^{-5}+2.382\times10^{-7}N^{0.8}}\right]\left[(\pi 0.008333\times7\times194.7N)\right]$$

$$= 1.706 \times 10^{8}$$

or

$$1.291 \times 10^{-4}(N/R)^{0.562} + 5.27 \times 10^{-5} + 2.382 \times 10^{-7}N^{0.8}$$

$$= \frac{\pi 0.008333 \times 7 \times 194.7N}{1.706 \times 10^{8}} \tag{51}$$

From equation 42,

$$(N/R)^{0.562} = 5.637N^{0.1974}.$$

Then, equation 51 becomes

$$7.280 \times 10^{-4}N^{0.1974} + 5.270 \times 10^{-5} + 2.382 \times 10^{-7}N^{0.8}$$

$$= 2.091 \times 10^{-7}N. \tag{52}$$

A trial and error solution of this equation yields a value of N of 31,700 tubes. This is to be compared with a value of 25,800 tubes for the same conditions when the secondary fluid is sodium.

In addition to comparing the effect of the secondary fluid on the number of tubes required, consideration must also be given to the pumping power requirements and the total weight of the tubes and their contained fluid. In making the calculations for the number of tubes required in the air radiator, both for the liquid alkali metals and for hydrogen, a fixed pressure drop was assumed for the air. Hence the power required to force the air through the air radiator is independent of the secondary fluid.

The pumping power required for the secondary fluid in the air radiator may be found in a manner similar to that used in the secondary heat exchanger; that is,

$$\bar{V} = \frac{M'}{\rho A_f} = \frac{47,400}{\rho C \times 500 \times (0.84D_o)^2(\pi/4)N} = \frac{171.1}{\rho C N D_o^2}. \tag{53}$$

For given conditions,

$$\bar{V} = 171.1/\rho \times 3.56 \times 31,700 \times 0.008333^2 = 21.83/\rho$$

$$\bar{V}^{1.8} = (21.83/\rho)^{1.8} = 257.3/\rho^{1.8}. \tag{54}$$

From equation 20,

$$\Delta P = \frac{0.092\rho^{0.8}(257.3/\rho^{1.8})7(1.316 \times 10^{-5})^{0.2}}{(0.084/12)^{1.2} \times 32.17 \times 144}$$

$$\Delta P = 1.456/\rho \text{ lb/sq in.} \tag{55}$$

$$\text{Pumping power} = \frac{A_f \bar{V} \Delta P}{550}$$

$$= \frac{\pi(0.084/12)^2 \times 31,700 \times 21.83 \times 1.456 \times 144}{4 \times 550\rho \times \rho}$$

$$\text{Pumping power} = 10.16/\rho^2 \text{ hp.} \tag{56}$$

Results for several pressures are shown in Table XIV.

TABLE XIV

PERFORMANCE OF AIR RADIATOR
SECONDARY FLUID HYDROGEN

Tube length = 7 ft   31,700 tubes

Pressure psia	ρ lb_m/cu ft	$\overline{V}$ ft/sec	ΔP psi	Pumping horsepower
400	0.04360	500.7	33.40	5314
700	0.07631	286.1	19.08	1744
1000	0.1090	200.3	13.36	854.6
2000	0.2180	100.1	6.680	213.7
4000	0.4360	50.07	3.340	53.14

These pumping powers are to be compared with that of 11.06 hp required for an air radiator operating under the same conditions, but using sodium as the secondary fluid.

The horsepower required for the secondary fluid may be reduced by reducing the tube length and thus increasing the number of tubes and the flow area. In the calculations which follow, the tube length is taken as 4 feet.

Substituting into equation 40,

$$3 = \frac{2.498R^{2.85}}{4^{1.85}N^{1.85}(0.1/12)^2\rho_m}$$

or

$$N = \left[\frac{2.498R^{2.85}}{3 \times 4^{1.85}(0.1/12)^2\rho_m}\right]^{1/1.85}$$

$$N = 40.05R^{1.541}/\rho_m{}^{1/1.85}. \qquad (57)$$

Using a density of 0.05078 lb/cu ft,

$$N = 200.5R^{1.541}. \qquad (58)$$

Substituting into equation 45,

$$h_o = \frac{0.46 \times 0.0393}{0.1/12}\left[\frac{(0.1/12) \times 1.3874 \times 10^6 R}{0.095(0.1/12)4N}\right]^{0.562}$$

$$h_o = 10{,}580(R/N)^{0.562}. \tag{59}$$

Then equation 44 reduces to

$$U_o = \frac{1}{\frac{1}{10{,}580}(N/R)^{0.562} + 5.27 \times 10^{-5} + 2.382 \times 10^{-7}N^{0.8}}. \tag{60}$$

Equation 43 reduces to

$$\left[\frac{1}{\frac{1}{10{,}580}(N/R)^{0.562} + 5.27\text{x}10^{-5} + 2.382\text{x}10^{-7}N^{0.8}}\right]\pi(0.008333)4N(194.7)$$
$$= 1.706 \times 10^8,$$

or

$$9.453 \times 10^{-5}(N/R)^{0.562} + 5.27 \times 10^{-5} + 2.382 \times 10^{-7}N^{0.8}$$
$$= \frac{\pi(0.008333)4N(194.7)}{1.706 \times 10^8} \tag{61}$$

From equation 58

$$(N/R)^{0.562} = 6.906N^{0.1974}.$$

Then

$$6.529 \times 10^{-4}N^{0.1974} + 5.27 \times 10^{-5} + 2.382 \times 10^{-7}N^{0.8}$$
$$= 1.195 \times 10^{-7}N, \tag{62}$$

A trial and error solution of this equation yields a value of N of 62,400 tubes. This is to be compared with a value of 45,200 tubes for the same conditions when the secondary fluid is sodium.

The method of determining the pumping power for the hydrogen is the same as previously used.

$$\bar{V} = \frac{171.1}{\rho C N D_o} = \frac{171.1}{\rho \times 3.56 \times 62{,}400 \times 0.008333^2}$$

$$= 11.09/\rho. \qquad (63)$$

$$\bar{V}^{1.8} = 76.03/\rho^{1.8}$$

$$\Delta P = \frac{0.092\rho^{0.8}(76.03/\rho^{1.8})4(1.316 \times 10^{-5})^{0.2}}{(0.084/12)^{1.2}32.17 \times 144}$$

$$\Delta P = 0.2453/\rho \text{ lb/sq in.} \qquad (64)$$

$$\text{Pumping power} = \frac{A_f \bar{V} \Delta P}{550}$$

$$= \frac{\pi(0.084/12)^2 62{,}400 \times 11.09 \times 0.2453 \times 144}{4 \times 550 \times \rho \times \rho}$$

$$\text{Pumping power} = 1.711/\rho^2 \text{ hp.} \qquad (65)$$

TABLE XV

PERFORMANCE OF AIR RADIATOR
SECONDARY FLUID HYDROGEN

Tube length = 4 ft 62,400 tubes

Pressure psia	ρ lb_m/cu ft	$\bar{V}$ ft/sec	ΔP psi	Pumping horsepower
400	0.04360	254.4	5.627	899.9
700	0.07631	145.4	3.215	293.9
1000	0.1090	101.7	2.251	144.0
2000	0.2180	50.88	1.125	36.0
4000	0.4360	25.44	0.5627	9.0

Weight of Tubes

Tube length = 7 ft

$$\text{Weight/tube} = (0.1^2-0.084^2)(\pi/4) \times 7 \times 12 \times 0.29$$

$$= 0.05633 \text{ lb}$$

Total weight of tubes = 0.05633 x 31,700 = 1786 lb.

This compares with a weight of tubes and the contained fluid of 1792 lb when the secondary fluid is sodium.

Tube length = 4 ft

$$\text{Weight/tube} = (0.1^2-0.084^2)(\pi/4)4 \times 12 \times 0.29$$

$$= 0.03219 \text{ lb}$$

Total weight of tubes = 0.03219 x 62,400 = 2008 lb.

This compares with a weight of tubes and the contained fluid of 1795 lb when the secondary fluid is sodium.

In determining the weight of the tubes when hydrogen is the secondary fluid, no increase in the tube thickness was made when high gas pressures were considered. It seems possible that suitable alloys can be developed and workmanship so perfected that tubes having the wall thickness used in these calculations will be satisfactory for pressures up to at least 1000 psia. Under these conditions it may be seen that, when long tubes (7 feet) are used, the total weight of the tubes and their contained fluid is substantially the same whether the secondary fluid is hydrogen or sodium. However, at 1000 psia, the hydrogen requires a pumping power of 854.6 hp to force it through the air radiator, but the sodium requires a pumping power of only 11.06 hp.

When shorter tubes (4 feet) are used, the weight of the tubes and their contained fluid is about 12 percent (200 lb) greater when the secondary fluid is hydrogen than when sodium is used. The use of the 4-foot tubes reduces the pumping power required for the hydrogen to 144 hp at a gas pressure of 1000 psia.

V. CONCLUSIONS

From the standpoint of system weight and pumping power requirements for the type of service assumed in this report, the best heat transfer gas (hydrogen) is not as satisfactory as the liquid alkali metals for secondary heat transfer fluids.

If present commercially available metals, such as stainless steel No. 316, are used for the tubes of the secondary heat exchanger, the air radiator, and the heat transfer piping, the system weight for a given pumping power will be many times greater when hydrogen is the secondary fluid than when an alkali liquid metal is used. It is recognized, however, that new alloys which have excellent strength characteristics at the elevated temperatures are being developed. Assuming that these alloys will become available and that workmanship in the manufacture of tubes and pipe can be improved, the weight of the tubes and the transfer pipe for hydrogen may be reduced to a value not greatly exceeding that required for the liquid alkali metals. In making this statement it is assumed that, because of corrosion possibilities, the tube and pipe walls must be made thicker, for a given pressure, when a liquid alkali metal is used in place of hydrogen.

It should be noted that no consideration has been given to the possible deleterious effect of the gas on the tubes. Certainly hydrogen may give difficulties, such as tube embrittlement, when used with common containing materials. It is difficult to predict whether or not special alloys may be developed which will be compatible with hydrogen or other gases which may be used in the place of hydrogen. If this is not possible, then the weight of the system with a gas as the secondary fluid will be more unfavorable than presented in this report.

When long tubes (7 feet) are used in the air radiator, the total weight of the tubes and fluid is substantially the same when hydrogen and liquid sodium are the secondary fluids, assuming the same tube wall thickness. The pumping power for hydrogen in the air radiator is excessive unless high pressures are used. (At 1,000 psia the pumping power for hydrogen is 854.6 hp.) Pumping powers for hydrogen may be decreased to small values by using shorter tubes (4 feet), but there will be an approximate increase of 12 percent in total weight.

The calculations presented in this report show that when using hydrogen as the secondary fluid the weight of the secondary heat exchanger can be made to approach that of an exchanger using a liquid alkali metal only by using very long tubes. When this is done, pumping power requirements will be higher for hydrogen unless gas pressures of at least 4,000 psia are used.

Unless new alloys are developed and workmanship during manufacture is improved, the weight of the transfer pipeline and its contained fluid for a fixed pumping power when hydrogen is used will materially exceed that when the secondary fluid is a liquid alkali metal.

In summary, it may be concluded that the liquid alkali metals are superior to gases as secondary heat transfer fluids as far as system weight and pumping power requirements are concerned, but the superiority of the alkali metals may be reduced to some extent when alloys are developed which will permit operation of the gas at pressures of at least 4,000 psia without an increase in the tube wall thickness.

Two other observations should be noted.

1. When the transfer pipe between the secondary heat exchanger is long, hydrogen is superior, as far as weight considerations are concerned, to heavy fluids such as bismuth and the lead-bismuth alloys.

2. For those fields of service where system weight and space are not critical factors, it is possible to design a secondary heat transfer system in such a manner that the pumping power requirements are not excessive for a gas such as hydrogen. Under these conditions, further consideration should be given to gases as secondary heat transfer fluids.

VI. BIBLIOGRAPHY

1. N. C. State College Report to ONR dated August 26, 1957. Title classified.

2. Interrelation of Factors Influencing Heat Exchanger Performance. N. C. State College Report to ONR dated October 15, 1958.

3. The Interrelation Between Various Alkali Metals and Characteristics of Secondary Heat Exchangers. N. C. State College Report to ONR dated December 1959.

4. Reid and Sherwood, The Properties of Gases and Liquids, McGraw-Hill Book Company, Inc., New York, 1958.

5. Ellenwood, Keelik and Gay, The Specific Heats of Certain Gases over Wide Ranges of Pressures and Temperatures, Cornell University Engineering Experiment Station Bulletin No. 30, 1942.

6. Doolittle, Doggett, and Martin, The Effect of Secondary Heat Transfer Fluids on Air Radiator Characteristics, Report to Office of Naval Research, U. S. Navy, December 1960.

7. McAdams, W. H., Heat Transmission, McGraw-Hill Book Company, Inc., New York, 1954.

8. Bowman, Mueller, and Nagle, "Mean Temperature Differences in Design," Transactions, ASME, 62, 283-294, 1940.